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Renewable and Sustainable Energy Reviews
8 (2004) 129–155
RENEWABLE
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Ejectors: applications in refrigeration technology

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Received 17 July 2003; accepted 13 October 2003

Abstract

This paper provides a literature review on ejectors and their applications in refrigeration. A number of studies are grouped and discussed in several topics, i.e. background and theory of ejector and jet refrigeration cycle, performance characteristics, working fluid and improvement of jet refrigerator. Moreover, other applications of an ejector in other types of refrigeration system are also described.

Keywords: Ejector; Refrigeration cycle; Jet refrigerator

Contents

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1364-0321/$ - see front matter
doi:10.1016/j.rser.2003.10.001

1. Introduction

Most industrial processes use a significant amount of thermal energy, mostly by burning fossil fuels. Some part of the energy released in combustion is inevitably rejected to the surroundings as waste. This waste heat can be utilized in certain types of refrigeration system such a jet refrigeration cycle.

With the use of the jet refrigeration cycle, the amount of electricity purchased from utility companies, required for conventional vapour compression in the refrigeration cycle, is reduced. Thus, utilization of the waste heat in refrigeration systems promotes mitigating the problems related to the environment, particularly by reduction of CO_2 emission from combustion of fossil fuels in boilers of utility power plants.

Although jet refrigeration systems seem to provide many advantages, vapour compression systems still dominate in all of the market sectors. In order to promote the use of jet refrigeration systems, further development is required to improve their performance and reduce the unit cost.

The aim of this paper is to provide basic background knowledge and a review of existing literature on ejectors and their applications in refrigeration cycles. The concepts and interesting points of each study were linked and grouped to other related studies and described as overview summaries. These differentiate this paper from other existing review papers [1,2]. It is hoped that this paper will be useful for any newcomer in this field of refrigeration technologies.

2. Background and ejector theory

The ejector, which is the heart of the jet refrigeration system, was invented by Sir Charles Parsons around 1901 for removing air from a steam engine's condenser. In 1910, an ejector was used by Maurice Leblanc in the first steam jet refrigeration system [3]. This system experienced a wave of popularity during the early 1930s for air conditioning large buildings [4]. Steam jet refrigeration systems were later supplanted by systems using mechanical compressors. Since that time, development and refinement of jet refrigeration system have been almost at a standstill as most efforts have been concentrated on improving vapour compression refrigeration systems.

A schematic view of a typical steam ejector is shown in Fig. 1. The flow process is also presented in Mollier's chart [4–12] Fig. 2. Referring to Fig. 1, as the high pressure steam (P), known as 'primary fluid', expands and accelerates through the primary nozzle (i), it fans out with supersonic speed to create a very low pressure region at the nozzle exit plane (ii) and hence in the mixing chamber. According to the differences of pressure of two positions, higher-pressure vapour, which, can be

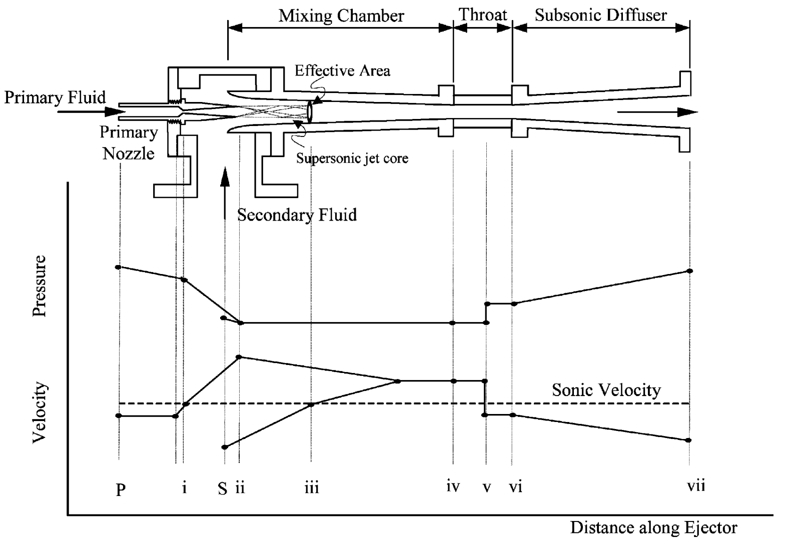


Fig. 1. Schematic view and the variation in stream pressure and velocity as a function of location along a steam ejector.

called the 'secondary fluid' (S), can be entrained into the mixing chamber. The primary fluid's expanded wave was thought to flow and form a converging duct without mixing with the secondary fluid. At some cross-section along this duct, the speed of secondary fluid rises to sonic value (iii) and chokes. This cross-section was defined by Munday and Bagster [13] as the 'effective area'. The experimental results

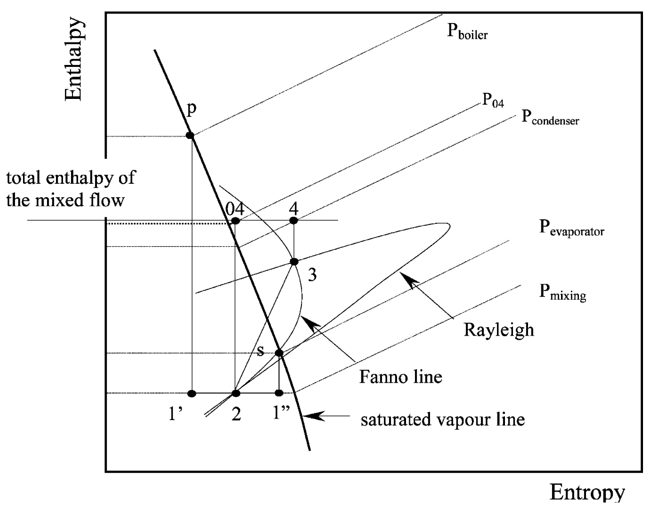


Fig. 2. Mollier's chart of an ejector.

and analysis, provided in [14,15], indicated that this hypothetical area was not constant but varied with the operating conditions. Munday and Bagster also suggested that the mixing process begins after the secondary flow chokes. This mixing causes the primary flow to be retarded whilst secondary flow is accelerated. By the end of the mixing chamber, two streams are completely mixed and the static pressure was assumed to remain constant [4] until it reaches the throat section (iv). The pressure in the mixing chamber was a function of primary fluid, secondary fluid and the back pressure of ejector [16]. Due to a high-pressure region downstream of the mixing chamber's throat, a normal shock of essential zero thickness is induced (v). This shock causes a major compression effect and a sudden drop in the flow speed from supersonic to subsonic. Please note that this normal shock is valid in the assumption of one-dimensional analysis only. In real situations, because of a thick boundary layer, the shock is not fully normal but includes complex oblique shock patterns. A further compression of the flow is achieved (vi) as it is brought to stagnation through a subsonic diffuser. The experimental results of static pressure profile measurement taken along the wall of ejector have shown these assumptions to be valid in the studies of Eames et al. [17], Huang et al.[14], Chunnanond and Aphornratana [18] and Chen and Sun [19].

The above one-dimensional ejector theory was first introduced by Keenan et al. [20]. Their mathematical analysis was based on an ideal gas dynamics together with the principles of mass, momentum, and energy conservation, and it has been used as a theoretical basis in ejector design for the past fifty years. However, Keenan's theory, cannot predict the constant-capacity characteristic that was proposed later by Munday and Bagster [13]. In order to eliminate the analytical error induced by the ideal gas assumption when the ejector issued with refrigerants, the thermodynamics properties of real gases were applied [8,21–25]. However, the studies of Aphornratana [25] and Abdel-Aal et al. [24] indicated that both approaches provide the similar results. Since the pressure in the mixing chamber is very low, therefore, the fluids behave like an ideal gas. Moreover, to make the simulated model become more realistic, the isentropic efficiency [8,10–12,25–35] including the friction losses [36–38] and pressure loss [39] were taken in to account. The values of these parameters widely differed within the range of 0.8–1.0, depending on ejector geometries and operating conditions [33,35,40].

Normally, the ejector design can be classified into two types according to the position of the nozzle. The ejector, which has the nozzle with its exit plane located within the suction chamber in front of the constant-are section, as described by Keenan's theory, the static pressure was assumed to be constant through the mixing process. Therefore, this kind of ejector is known as a 'constant-pressure mixing ejector' (Fig. 3a). For the nozzle with its exit located within the constant-area section, the ejector is called a 'constant-area mixing ejector' (Fig. 3b) [20,41,42]. At the beginning, it was thought that a constant-area mixing ejector could entrain a higher amount of secondary flow than a constant-pressure mixing ejector [43]. Both types of ejector have been extensively tested experimentally over the intervening years. It was found that the constant-pressure mixing ejector had a better per-

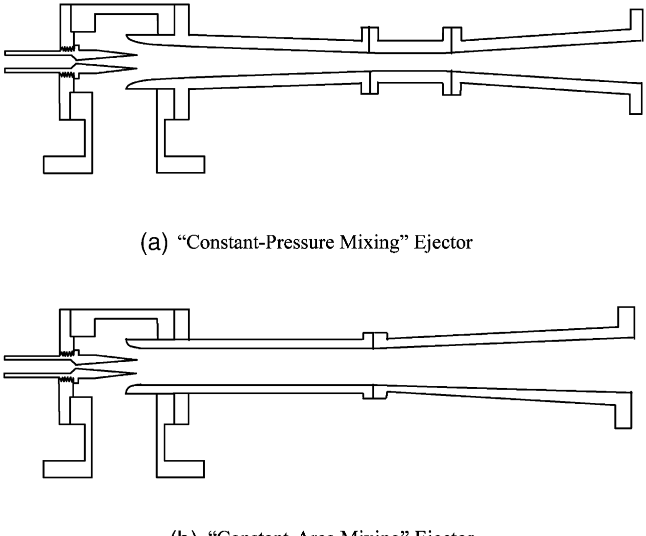


Fig. 3. Schematic view of an ejector. (a) constant-pressure mixing ejector; (b) constant-area mixing ejector.

formance than the constant-area one. Therefore, almost all current study has been focused on the constant-pressure mixing ejector.

According to Chou et al. [44], the choking phenomenon that took place in an ejector can be categorized into three types regarding to their flow and mixing characteristic. For two-phase fluid, typically used in refrigeration application, slightly of the mixing process was thought to start since the secondary flow was entrained and accelerated through the primary fluid expanded wave. The large difference in flow speed produced the 'shear stress layer' interface between two contact streams and shear mixing occurred. Therefore, the choking was taking account for the mixed stream not only for the secondary fluid, as proposed by Munday and Bagster [13]. This assumption was applied in the mathematical simulation of Chou et al. [44]. The calculated performance was compared to the experimental results of R-113 [14], R-141b [15] and steam [37] refrigerators. Moreover, the error of a predicted performance was claimed to reduce when compared to the conventional 1-D ejector theory. The experimental analysis of flow visualizing in an ejector of Desevaux [45] can be used to support the idea of Chou et al. [44]. The laser induced fluorescence, laser tomography method including image processing were used to investigate the mixing zone in an ejector. However, it should be noted that the test was not carried out directly in a refrigeration application. Constant-area mixing ejector was used with entire supersonic regime.

Not only the experimental investigation, [17–19], but some researchers tried to explain the flowing and mixing processes through an ejector by using the Computational Fluid Dynamics (CFD) software package. Riffat et al. [46] analyzed the 3-

D model of an ammonia ejector in 1996. At that time, the governing equation responsible for the compressible flow was too complicated. Therefore, the simulated incompressible flow ejector cannot be applied to the practice flow as well. Recently, with the rapid development of the computer and its resources, Rusly et al. [42] simulated 3-D flow inside the R245 ejector. In this study, the compressible real gas model was applied on the large numbers of grid elements. The results provide the reliable simulated insight observation of the flow process happening inside an ejector, including the existence of the expanded converging duct of primary fluid and the thermodynamics shock wave.

There are several parameters used to describe the performance of an ejector. For refrigeration applications, the most important parameters are 'an entrainment ratio' [14] and 'a pressure lift ratio':

$$\text{entrainment ratio, Rm} = \frac{\text{mass of secondary flow}}{\text{mass of primary flow}} \tag{1}$$

$$\text{pressure lift ratio} = \frac{\text{static pressure at diffuser exit}}{\text{static pressure secondary flow}} \tag{2}$$

The entrainment ratio is related to the energy efficiency of a refrigeration cycle and the pressure ratio limits the temperature at which the heat can be rejected [47]. Therefore, there is no doubt that an ejector operating at the given operating conditions with the highest entrainment ratio and maintaining the highest possible discharged pressure will be the most desired ejector.

From the Mollier's chart shown in Fig. 2, it can be seen that the normal shock, which creates a major compression effect, causes loss in total pressure of the mixed stream. If the mixed stream is brought to stagnation state isentropically (without a normal shock), the exhaust pressure will be as high as P_{04}. This can be considered as an ideal ejector, which can be considered as an isentropic compressor driven by an isentropic turbine as shown in Fig. 4a. It must be noted that this model is not a reversible system even if the loss due to the shock is eliminated. Another loss caused by the mixing of two fluid streams (primary and secondary flows) remains. Not only the shear mixing, but the shear force was also introduced to the flowing process by the shear stress layer. These two factors were considered as the cause of entropy generation, and hence, the irreversibility of an ejector. A reversible ejector model which eliminates all losses is shows in Fig. 4b, both isentropic turbine compressor discharge at the same entropy and back pressure.

The idea of designing an ejector with minimum losses from normal shock was proposed and named 'the Constant Rate of Momentum Change method, (CRMC)', by Eames [48]. The ejector was designed so that the static pressure of the mixed flows was allowed to gradually increase from entry to exit while passing through the ejector. Without the shock, therefore, there is no stagnation pressure loss. It was claimed that the pressure lift ratio of the CRMC ejector was increased while the entrainment ratio remained unchanged. In the same manner, the losses created by normal shock were minimized in the experiment of Chunnanond and Aphornratana [49]. The method was to reduce the flow speed to the sonic value by

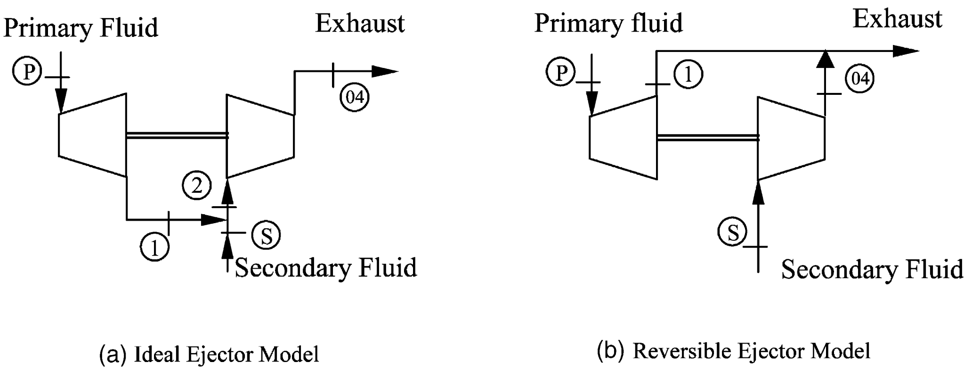


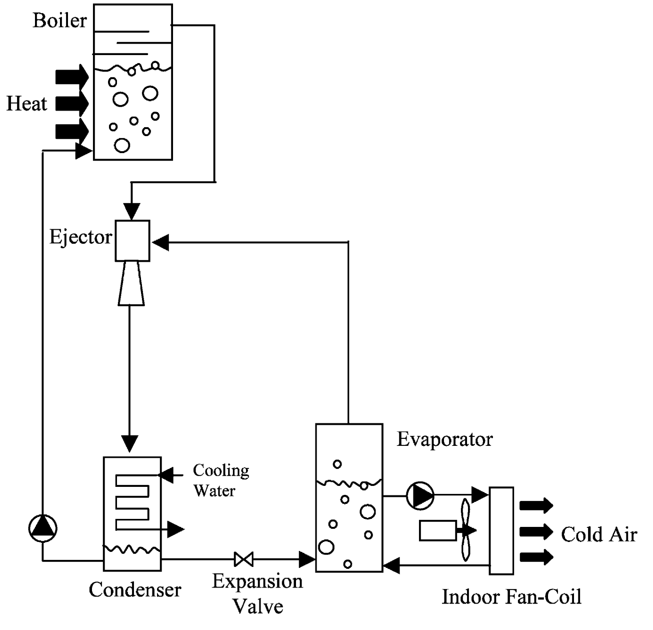
Fig. 4. Schematic view of ejector compared to turbo machinery equipment (working state referred to Fig. 2). (a) ideal ejector model; (b) reversible ejector model.

using the ejector with smaller throat section. The experimental results concurred with Eames [48]. However, it was found that the ejector lost it function if the throat diameter was too small.

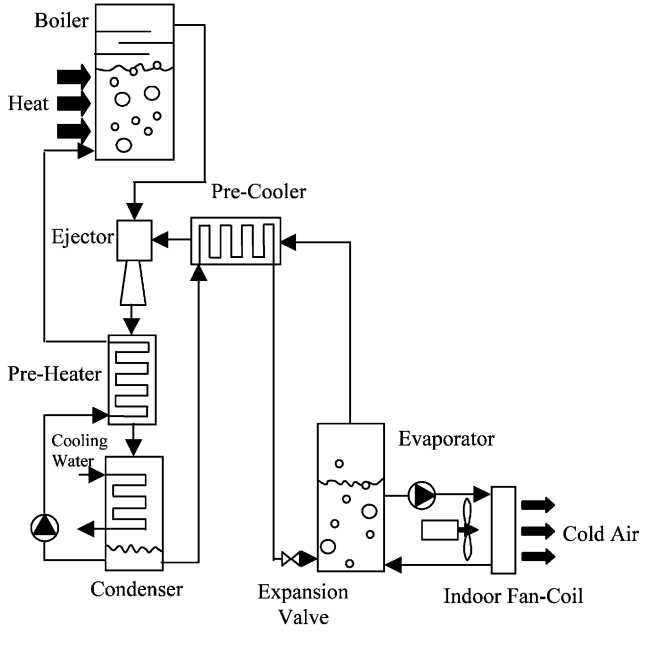
In the study of Garris [50], the primary fluid was proposed to decelerate and expand through the self-rotating skewed primary nozzle of a novel designed ejector. It was thought that the tangential motion of primary fluid could reduce the shear force and, consequently, the shear mixing of the entrainment process. However, this design configuration introduced some moving parts and it was extremely complicated when compared to the conventional ejector. Similar to Garris [50], but simpler, the irreversibility produced by the mixing process was also reduced by using the petal nozzle instead of conventional conical primary nozzle [51]. In their experimental study, it is postulated that the more efficient mixing in ejector can be achieved by using a petal nozzle, and then more energy saved can be recovered to the back pressure. The ejector could operate with higher pressure lift ratio. Consequently, other characteristics of an ejector with petal nozzle are similar as those of conical nozzle.

3. Jet refrigeration cycle

Fig. 5a shows a schematic diagram of a jet refrigeration cycle. A boiler, an ejector, and a pump are used to replace the mechanical compressor of a conventional vapour compression refrigeration system. As heat is added to the boiler, the high pressure and temperature refrigerant vapour is evolved and used as the primary fluid for the ejector. The ejector draws low pressure refrigerant from the evaporator as its secondary fluid. This causes the refrigerant to evaporate at low pressure and produce the useful refrigeration. The ejector discharges its exhaust to the condenser where it is liquefied at the ambient temperature. Part of the liquid refrigerant is pumped back to the boiler whilst the remainder is returned to the evaporator via a throttling device. Often, the operating condition of boiler, evapor-



(a) Typical Jet Refrigerator



(b) Jet Refrigerator with a Pre-Cooler and a Pre-Heater

Fig. 5. Schematic diagram of a jet refrigeration cycle. (a) typical jet refrigerator; (b) jet refrigerator with a pre-cooler and a pre-heater.

ator and condenser of a jet refrigeration cycle are defined by heat source, refrigerated purpose and local climate respectively. The input required for the pump is typically less than 1% of the heat supplied to the boiler, thus, the actual COP [28] may be given as:

$$\mathrm{COP} = \frac{\text{refrigeration effect at the evaporator}}{\text{heat input at the boiler}} \tag{3}$$

In many studies [12,14,22,23,52–57] the pre-cooler and the pre-heater (Fig. 5b) are installed to the conventional system, in order to improve the system efficiency. The temperature of the refrigerant from the condenser is slightly increased and decreased before entering the boiler and evaporator, respectively. Therefore, the required heat input and the cooling load of the system are reduced.

Apart from using the mechanical driven pump, there were two alternate ways to return the liquid refrigerant to the boiler. The first alternative was proposed by Riffat and Holt [32]. The principal of heat pipe is applied to an ejector. The condensed liquid refrigerant from the condenser returned through the wick by capillary action to the boiler. Another practical alternative was achieved by using the gravitational head difference between a condenser and a boiler [11,58]. The requirement of the recirculation pump is eliminated. Thus, the system becomes the purely heat-operated cycle.

3.1. Performance characteristics

The performance of a jet refrigeration system is directly dependent on the efficiency of the used ejector. The entrainment ratio is the function of the operating conditions and the ejector geometries. Moreover, the entrainment ratio is limited by the ejector critical back pressure. The experimental investigations of ejectors and their application on refrigeration cycles have been presented [14–19,25,28,51,56–60]. The typical system's performance curve for the specified boiler and evaporator pressures is shown in Figs. 6 and 7. There are three regions: choked flow in the mixing chamber, unchoked flow in the mixing chamber, and reversed flow in the mixing chamber.

At the condenser pressure below the 'critical value' [14], the ejector entrains the same amount of secondary fluid. This causes the cooling capacity and COP to remain constant. This phenomenon is thought to be caused by the flow choking within the mixing chamber, [13]. When the ejector is operated in this pressure range, a transverse shock, which creates a compression effect, is found to appear in either the throat or diffuser section. The location of shock process varies with the condenser back pressure [14,17–19,22,24,62,63]. If the condenser pressure is further reduced, the shock will move toward the subsonic diffuser and vice versa. When the condenser pressure is increased higher than the critical value, the transverse shock tends to move backward into the mixing chamber and interferes with the mixing of primary and secondary fluid. The secondary flow is no longer choked, thus, the secondary flow varies and the entrainment ratio begins to fall off rapidly.

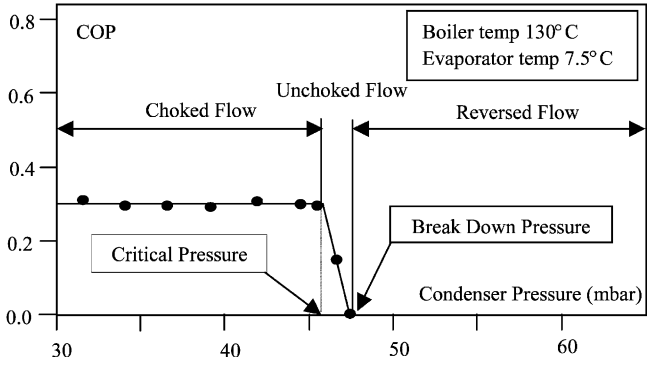


Fig. 6. Performance of a steam jet refrigerator based on experimental data provided by Aphornratana [25].

If the condenser pressure is further increased, the flow will reverse back into the evaporator and the ejector loses its function completely.

For the condenser pressure below the critical value, the mixing chamber is always choked. The flow rate of the secondary flow is independent from the downstream (condenser) pressure. The flow rate can only be raised by an increase of the upstream (evaporator) pressure. The critical condenser pressure is dependent on the momentum and pressure of the mixed flow. The pressure is equal to that of the evaporator. As the secondary flow enters the ejector at low speed, momentum of the mixed flow is equal to that of the primary fluid exiting from the primary nozzle. Thus, to increase the critical condenser pressure, the pressures at the boiler or the evaporator must be increased.

A decrease in the boiler pressure causes the primary fluid mass flow to reduce. As the flow area in the mixing chamber is fixed, an increase in the secondary flow results. This causes the cooling capacity and COP to rise. However, this causes the momentum of the mixed flow to drop. Thus, the critical condenser pressure is

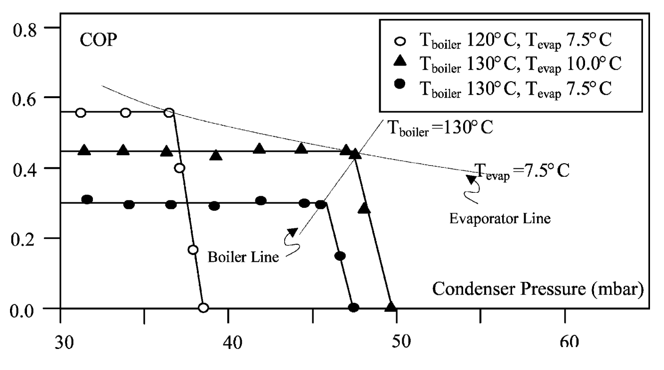


Fig. 7. Effect of operating temperatures to performance of a steam jet refrigerator based on data provided by Aphornratana [25].

reduced. On the other hand, an increase in the evaporator pressure, which is the ejector's upstream pressure, will increase the critical condenser pressure. This also increases the mass flow through the mixing chamber. Thus, increases of cooling capacity and COP result. Even though raising evaporator pressure would help to increase the entrainment ratio, this would sacrifice the desired cooling temperature.

According to the performance characteristic, it is desirable to operate the ejector at critical condenser pressure. Fig. 8 shows an example of the effect of operating conditions and a performance map [14,19,25,28,51,59,61–63] of a steam jet refrigerator. A point on the diagram shows the system's COP when the ejector is under critical condenser.

Not only the operating conditions, but the ejector geometry was found to affect the ejector performance [5,6,17,18,20,26,33,36,37,43,49,51,59,67]. In the study of Keenan et al. [20,43], Hoggarth [36], ESDU [67], Eames et al. [17] and Aphornratana and Eames [59], it was found that the ejector performance, i.e., cooling capacity, COP and critical condenser pressure of a jet refrigerator can be varied by changing the position of primary nozzle. Retracing the nozzle into the mixing chamber causes the cooling capacity and COP to increase with the expenses of critical condenser pressure. According to their tests, a single optimum nozzle position cannot be defined to meet all operating conditions. Moreover, the optimum nozzle position found by Aphornratana and Eames [59] was in contrast to the recommendation of ESDU [67]. It was thought that each ejector required a particular optimum nozzle position. The experimental study of the effect of primary nozzle throat size and ejector geometry on system performance was conducted [17,18,61]. The influence of using small primary nozzle throat was similar to that of decreasing boiler saturation pressure whilst the influence of the primary nozzle exit diameter was not significant [17]. It was verified that the critical condenser pressure can increase when utilizing: (1) a longer and larger mixing chamber and (2) a longer and smaller diameter throat section.

The previous explanations indicate the influences of geometry of an ejector on its performance and the constraint of a fixed-geometry ejector refrigeration system in

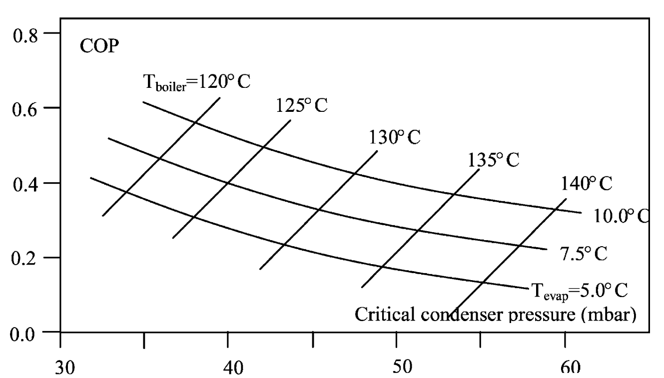


Fig. 8. Performance map of a steam jet refrigerator based on data provided by Aphornratana [25].

achieving the optimum performance under various operating points. Sun, [26] proposes the utilization of variable-geometry ejector (ejector which can change its geometries) in the refrigeration application. The optimization of 1-D mathematical model comprises with an ejector design guide of ESDU [67] provide the ejector sizes in the given range of operating conditions. Their computed results showed the elimination of constant-capacity characteristic, thus, the system was more flexible in operation. The practical solution, the multi-ejector refrigeration system, was proposed in the series of studies of Sokolov and Hershgal [22,23,52,68]. The parallel array of ejectors is placed between a common inlet and the condenser. The selector switches were used to select the most appropriate ejector for each operating condition. The multi-ejector can also be arranged in series [5]. The series arrangement is suitable for the system where there is a large temperature difference and thus, a high pressure lift ratio is required. Grazzini and Mariani [69] proposed the mathematical model used to design the series-ejector. Their results of a two-stage ejector showed a 15% increase of pressure lift ratio while the entrainment ratio was unchanged.

In the experimental study of Al-Khalidy [6,35,70] and Aphornratana et al. [71], the performance characteristics were different. Choking of the secondary flow in the mixing chamber was not presented. The secondary mass could be increased with the decreasing of condenser pressure and higher incoming motive mass. Therefore, the cooling capacity and COP were always increased when the condenser pressure dropped, the boiler pressure increased and evaporator pressure increased. In addition, when there was no choking in the mixing chamber, the system was more flexible, the ejector was more efficient and the system could provide a better performance. Unfortunately, there are no certain criteria to design and construct this kind of ejector.

3.2. Working fluids

Performance of a jet refrigeration cycle is dependent on thermodynamic properties of the working fluid. Table 1 lists some fluids commonly used for experimental studies. The following requirements should be met:

- The fluid should have a large latent heat of vaporization in order to minimize circulation rate per unit of cooling capacity.
- The fluid pressure at the boiler temperature should not be too high in order to avoid heavy construction of the pressure vessel and to minimize the power required by pump [6].
- The fluid should be chemically stable, non-toxic, non-explosive, non-corrosive, environmental friendly and low cost [6].
- Transport properties that influence heat transfer, e.g., viscosity and thermal conductivity should be favorable.
- Working fluid with smaller value of molecular mass requires comparatively larger ejectors for the same system capacity. The difficulties of constructing small scale ejector components should be considered [72]. However, higher molecular mass fluid leads to an increase in entrainment ratio and ejector efficiency [73]

Table 1
Fluids for a jet refrigerator[a]

	R-11	R-12	R-113	R-123	R-141b	R-134a	R-718b (water)
Boiling point at 1 atm (°C)	23.7	−29.8	47.6	27.9	32.1	−26.1	100.0
Pressure at 100 °C (kpa)	824	3343	438	787	677	3972	101
Molecular mass (kg/kmol)	137.38	120.92	187.39	152.93	116.9	102.03	18.02
Latent heat at 10 °C (kj/kg)	186.3	147.6	155.3	176.8	129.4	190.9	2257.0
Global warming potential (GWP)	1	3	1.4	0.02	0.15	0.26	0
Ozone depletion potential (ODP)	1	0.9	0.8	0.016	0	0.02	0
Wet/dry vapour	Wet	Wet	Dry	Dry	Dry	Wet	Wet

[a] GWP scale range from 0 to 1 (for CO_2, GWO = 1). ODP scale range from 0 to 1 9 (for R11, ODP = 1).

- According to Chen et al. [61], working fluid for a jet refrigerator can be categorized as wet vapour and dry vapour as shown in Fig. 9. For wet vapour fluid, its saturated vapour line forms a negative slope in the *T–s* diagram. For dry vapour fluid, there is no phase change during the expansion process through the primary nozzle. On the other hand, for wet vapour fluid, small droplets may be formed at

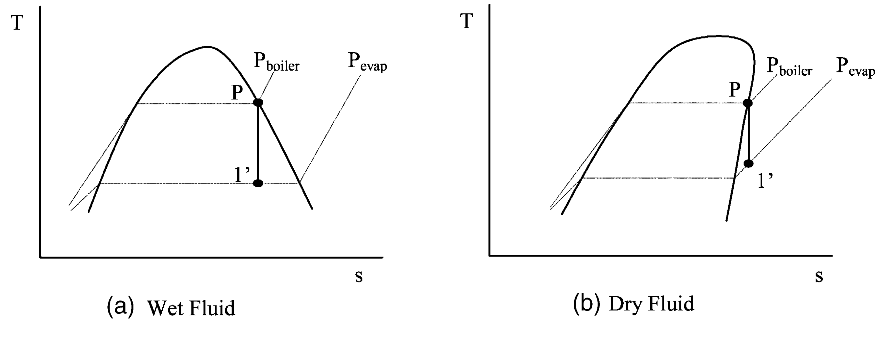


Fig. 9. Expansion process of refrigerant through the primary nozzle.

the nozzle exit, that block the effective are and bump into the wall and cause damage [5,61]. This may be eliminated by superheating the fluid before entering the nozzle. Dry vapour is more desirable than wet vapour fluid. However, the use of superheated motive steam causes a slight decrease in ejector efficiency [5,21].

Using water as the working fluid for a jet refrigerator provides many advantages. Its extremely high heat of vaporization causes a low circulation rate for given cooling capacity. Therefore, low mechanical power is required for the pump. Water is inexpensive and has minimal environment impact (zero ozone depletion and global warming potential). However, there are some drawbacks. Using water as a refrigerant limits the cooling temperature to above 0 °C and the system must be under vacuum condition. Moreover, water has very large specific volume at typical evaporator conditions and to minimize the pressure loss, pipe diameter must be large to accommodate the large volume flow rate [19]. Experiments show that a steam-jet refrigerator requires a boiler temperature between 120 and 160 °C. The system requires relatively low condenser pressure, thus, a water-cooled condenser is a must [25]. Thus, with water as a refrigerant, the useful range of operating temperature is thermodynamically restricted.

Experimental study by Holton [73] shows that an ejector performs better with high molecular weight fluid. Many attempts have been made by using various halocarbons-based refrigerants (CFCs, HCFCs and HFCs) such as R-11 [71,74,75], R-12 [16], R-113 [6,12,14,25,70], R-114 [23,52,56], R-123 [53], R-134a [19], R-141b [15,33,37,54,55,76], R-142 [77], R717 [7]. An advantage of jet refrigerator system using halocarbon compound is that low grade thermal energy as low as 60 °C can be used [1,39].

Sun [72] simulated the jet refrigerator with various refrigerants including, water, R11, R12, R13, R21, R123, R142b, R134a, R152a, RC318 and R500. The study shows that, R12 has the highest entrainment ratio and COP. However, R12 is CFC and its use will soon be prohibited. It was noticed that, when the simulated operating conditions were proper adjusted, COP of the water system was increased and it became a serious competitor to others. Another comparative simulation study was carried out by Cinzungu et al. [38]. In their study, four environmental friendly refrigerants, R-123, R-134a, R-152 and R717, were involved. The constructed model was first calibrated with the experimental results [66]. It was found that, with the same ejector, R-134a and R-512a were suitable for 70–85 °C heat source and ammonia was suitable for the heat source whose temperature was greater than 90 °C.

Based on experimental results provided in the literature, jet refrigeration systems based on halocarbon refrigerants seem to be more practical than systems using water. They can provide cooling temperature below 0 °C and require relatively low boiler temperature even in a higher environment (condenser) temperature. Thus the system is possible to be applied with a simple flat-plate solar collector, automobile exhaust gas [12,74,75] and use an air-cooled condenser, while producing am accept-

able COP and cooling temperature. These are impossible for the steam–water system.

4. Solar jet refrigeration system

One significant characteristic of the jet refrigeration system is its utilization of low temperature thermal energy. Free solar energy harnessed by the solar collector is quantitative and qualitative enough to generate the motive fluid for an ejector refrigeration system. However, according to the characteristic of an ejector and the initial investment cost, the solar jet refrigerator system is suitable for the application of air-conditioning system rather than that of the refrigeration purposes. The solar jet air-conditioning system can cope with the availability of energy source required to remove the increased cooling load, incoming with higher solar intensity. Until now, the 7 kW-prototype machine is installed and tested with the office building in the UK [58], while, the construction plan of the 13 kW machine for the hospital air-conditioning system is provided in [57].

A solar jet refrigerator consists of two sub-systems, which are the ordinary refrigeration system and the solar system, Fig. 10a. Heat collected by a solar system can be delivered and drive the refrigeration system by several options. For an ideal system, the solar cycle acts as the boiler of the refrigeration system. The refrigerant is forced and passes through the absorber of solar collector. This system is not desirable because leaks will cause damage, the pump should be sized according to the required flow rate and system pressure and controlling of the boiler pressure become more difficult [78]. To eliminate this restriction, in practice, the solar and refrigeration systems are separated, Fig. 10b. Heat from the solar collector is carried by the intermediate medium and transferred to the refrigerant by the boiler heat exchanger. The heat transferring mediums should have the boiling point higher than the possible temperature in the system, low viscosity and a good heat transfer property. Water with a corrosion inhibitor additive and transforming oil are recommended for operating temperature below and above 100 °C, respectively. In order to prevent the failure of a solar system (i.e. from the uncertainty of the weather), an auxiliary backup heater, driven by gas or oil, may be added to the storage tank or the exit of the solar collector to ensure the stability of boiler temperature [11,54,78,79].

The overall efficiency of the solar jet refrigeration cycle can be expressed [11,54,55,68] as:

$$\mathrm{COP}_{\mathrm{overall}} = \eta_{\mathrm{solar}} \times \mathrm{COP}_{\mathrm{jet\ refrigerator}}$$

It is obvious that not only the performance of the refrigeration system itself, but the thermal efficiency of solar collector is also the parameter affecting on the overall performance of solar jet refrigerator. According to the above relation, the mathematical model used to predict the overall system performance is therefore always presented in two parts, the prediction of ejector refrigerator and solar collector performance, respectively [11,54,55].

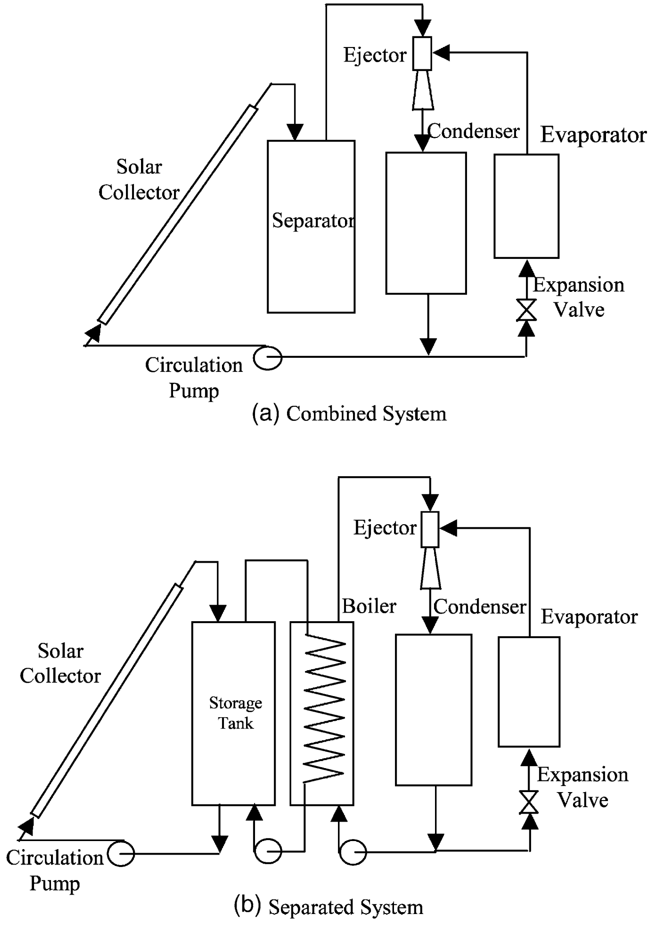


Fig. 10. Schematic diagram of solar jet refrigeration system. (a) combined system; (b) separated system.

The efficiency of a solar system depends on the collector type, solar radiation intensity, and the system operating conditions. When the boiler of the refrigeration system's operated at the temperature between 80 and 100 °C, a single glazed flat-plate type collector with a selective surface is recommended [11,24,15]. The expensive vacuum tube [55,58] and parabolic solar concentrating [57,78,79,81] collector can provide higher thermal efficiency when a higher operating temperature is required. Not only is the efficiency of a solar system but the economic analysis [2,55,58] also needed for the solar collector selection. The installation of a very high efficiency solar collector may give the significant increase of overall efficiency. On the other hand, the unit cost per watt of cooling is also increased and the break-even point may not be met. The mathematical models used to investigate the effect of solar collector types on the system performance were given in [11,54,55].

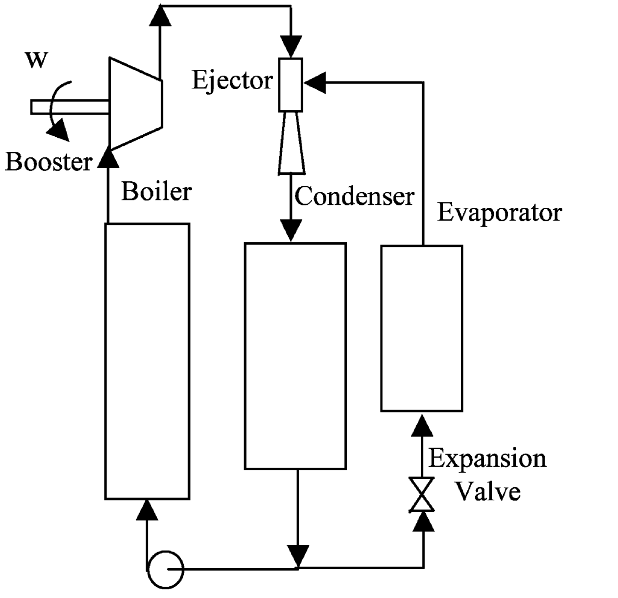
5. Compression enhanced jet refrigeration cycle

The performance of the jet refrigeration cycle refers to the entrainment ratio and the critical condenser pressure of an ejector. The only way to increase the secondary flow simultaneously with the ejector discharge pressure is to increase the secondary pressure. Unfortunately, for the conventional jet refrigerator, increasing evaporator temperature (refrigeration effect) is the must. In 1990, Sokolov and Hearshgal, [21] introduced new configurations of efficient uses of the mechanical power in order to enhance the secondary pressure without disturbing the refrigeration temperature, which are: (1) the booster assisted ejector cycle (Fig. 11a) and (2) the hybrid vapour compression-jet cycle (Fig. 11b). Their simulated results show that the compression enhanced ejector can significantly improve the system performance. Please note that, the power required by the booster is much greater than that required by the circulation pump and cannot be omitted in the evaluation of system performance.

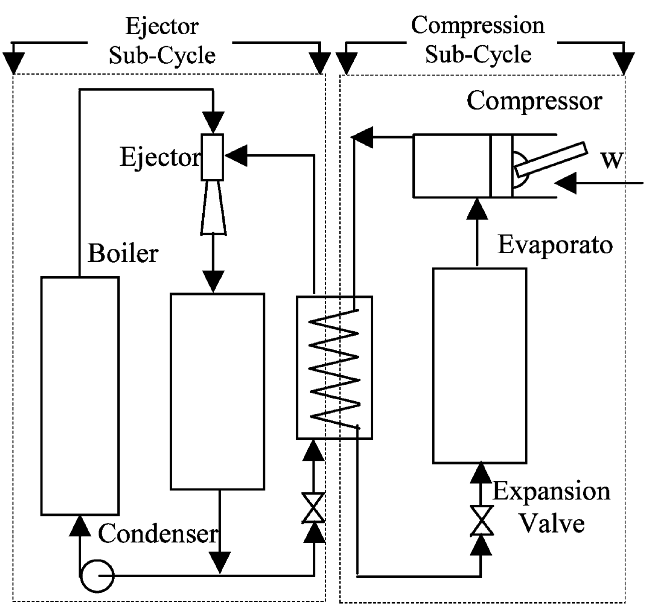
The booster assisted ejector cycle [4,21,77] is very similar to the conventional ejector cycle. The low-pressure ratio mechanical-driven compressor is placed between the evaporator outlet and ejector suction line. Therefore, the ejector can sense a higher suction pressure, which provides an increase of its performance. Anyway, in practice, the entrainment of oil droplets from the booster may adversely affect the smooth and clean operation of the ejector. The reduced compression ratio of ejector allows the application of low temperature refrigeration to the solar driven jet refrigeration system.

The hybrid compression-jet refrigeration system [21–23,29,52,56,80] consists of a conventional compression and ejector sub-cycles with heat exchanger as an interface between them. According to its configuration, pressure ratios across the ejector and the compressor are maintained at low level. The heat load is transferred from an evaporator to a heat exchanger and then compressed and rejected to the surrounding at a condenser. In other words, the ejector sub-cycle was used as the heat rejection system. If a single refrigerant is used, the heat exchanger is replaced by the mixing chamber and combined both heat and mass transfer processes. The concepts and design procedures of the system were explained in [22], while the description of the constructed multi-ejector R114 machine and its experimental data were given in [23]. In [52], the mathematical simulation (effect of operating condition) of a constructed machine and a machine with the utilization of solar energy were made respectively. However, R114 was found to be harmful to the environment and be prohibited, Da-Wen Sun [29] conducted the mathematical simulation of an environmental friendly solar system. Steam and R134-a were used as the refrigerant in an ejector and a compression sub-cycle, respectively. The simulated results showed that the COP of the system could be improved up to 50% compared to the conventional vapour compression system. The analysis of maximum possible COP of a solar powered hybrid compression-jet refrigerator, in term of Carnot efficiency, was provided in [80].

It is obvious that the booster and the vapour compression cycle can provide higher thermal efficiency to the jet refrigeration system. On the other hand, either



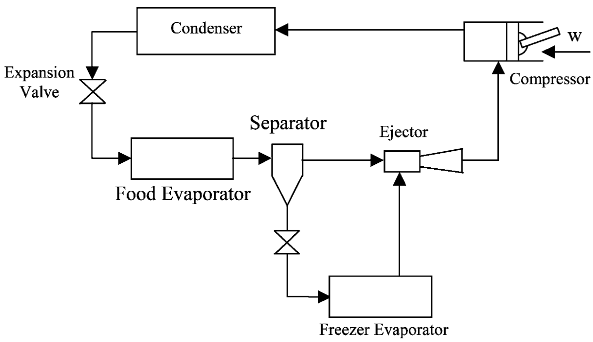
(a) Booster Assisted Ejector Refrigeration Cycle



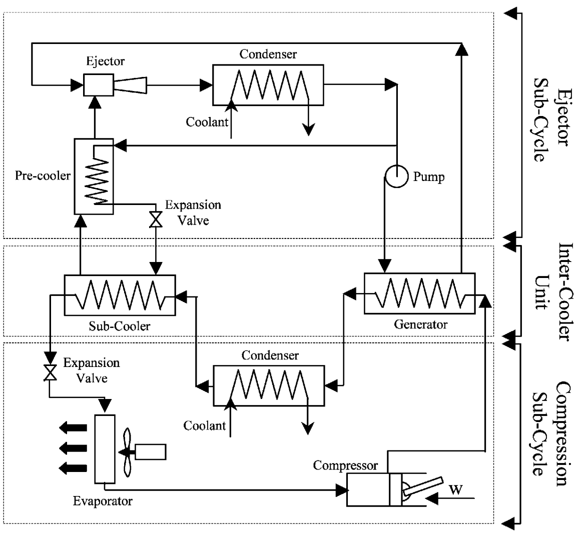
(b) Hybrid Compression-Jet Refrigeration Cycle

Fig. 11. Schematic diagram of compression enhanced jet refrigeration cycle. (a) booster assisted ejector refrigeration cycle; (b) hybrid compression-jet refrigeration cycle.

installation of an ejector [27] or its refrigeration system [76] also can improve the thermal efficiency of the ordinary vapour compression refrigerator. Tomasek and Radermacher [27] propose the use of ejector in the domestic fridge-freezer. Usually, a standard household fridge-freezer is equipped with two evaporators (fresh food and freezer compartment) and only one compressor. The required electricity used to compress the refrigerant from the freezer evaporator to the condenser level is high. In Fig. 12a, the installation of ejector reduces this pressure different, thus, less specific work and the system COP is improved. The calculated performance of ejector-combined system was compared to that of single-compressor (original) and



(a) Proposed by Tomasek and Radermacher, [27]



(b) Proposed by Huang et al, 2001 [76]

Fig. 12. Schematic diagram of the combined ejector-compression cycle. (a) proposed by Tomasek and Radermacher, [27]; proposed by Huang et al, 2001 [76].

two-compressor (dual-loop) system. Even though the ejector-combined system claimed 12.4% performance improvement from the single-compressor system, the dual-loop system has the highest COP improvement.

It is known that the increasing of sub-cooling level before entering the expansion device can increase COP of the conventional vapour compression system. Moreover, a plenty amount or heat is rejected as wasted from the very high temperature refrigerant of the compressor outlet. The combined ejector-compression machine of Huang et al. [76] allows the wasted heat from the superheated vapour refrigerant in the compression sub-cycle to drive the ejector sub-cycle (Fig. 12b). The cooling effect obtained from the ejector-cooling device is used to cool the liquid condensate of compression sub-cycle to a sub-cooled state to increase the COP of the system. Their analytic and experimental results verified the feasibility of the system. The COP of the system was, on average, 10% improved and a maximum of 24% improvement was claimed.

6. Hybrid ejector-absorption refrigeration cycle

There were several approaches used to improve the performance of a single-effect absorption refrigeration system. Many literatures were reviewed and provided by Srikhirin et al. [81]. Applying ejector to the conventional absorption system is one of the remarkable alternatives. The appropriate installation configuration introduces the magnificent improvement of COP to nearly to that of a typical double-effect absorption cycle machine. Moreover, according to the simplicity of the hybrid ejector-absorption refrigeration machine, its capital investment cost is comparatively low when compared to other conventional high performance absorption cycle systems.

Many researchers have attempted to improve the performance of absorption systems by using an ejector to raise the absorber pressure to a level higher than that in the evaporator and, consequently, to reduce the solution concentration. Kuhlenschmidt [82] proposed the development of an absorption system using working fluid based on salt-absorbent. This system utilized a two-stage generator similar to that used in a double-effect absorption system, Fig. 13. The low-pressure refrigerant from the second-effect generator is used as a motive fluid for the ejector and entrains vapour refrigerant from the evaporator. Therefore, the concentration of solution within the absorber can kept from crystallization when the system is required to operate with low evaporator temperature or with high absorber temperature. Neither theoretical nor experimental results of this system are available yet. However, it can be expected that the COP of the system will not be higher than that of a single-effect absorption system, as some of the vapour refrigerant generate discharged directly to the absorber (as the motive fluid) without producing any cooling effect.

In contrast with Kuhlenschmidt, Chung et al. [83] and Chen [84] elevated the absorber pressure by using a high-pressure liquid solution returned from the generator as the ejector's motive fluid (Fig. 14). Experimental investigation showed that,

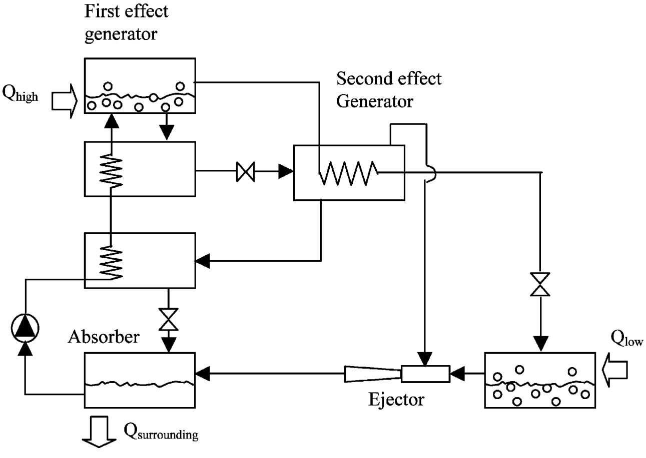


Fig. 13. Schematic diagram of the hybrid ejector-absorption cycle by Kuhlenschmidt [82].

by using DMETEG/R22 and DMETEG/R21 as working fluids, the pressure ratio between the absorber and the evaporator of 1.2 were found. The increasing in absorber pressure results in the circulation of the solution being reduced lower than that for a conventional system operated at the same condition. Thus, an improvement of COP can be expected. The mathematical simulation of the similar system, for heat pump application, was conducted by Shi et al. [85]. Unfortunately, the inappropriate working fluids, $LiBr/H_2O$ were selected. A liquid-driven ejector is not suitable to operate with low-density vapor such water. Only high-pressure and high-density refrigerant can be used. Therefore, the calculated results showed that the cooling capacity at the evaporator and thus the COP of the system were almost unchanged.

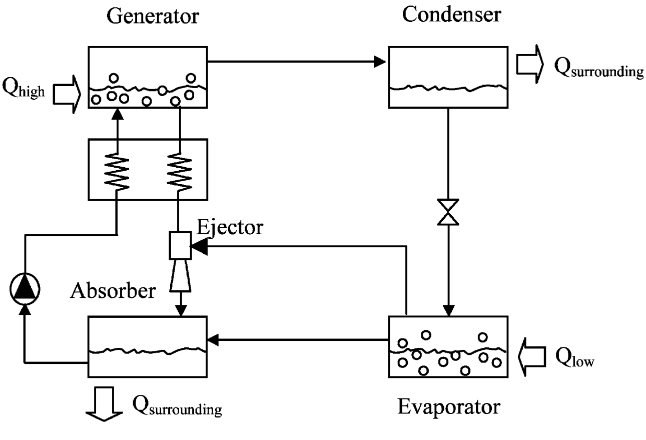


Fig. 14. Schematic diagram of the hybrid ejector-absorption cycle by Chung et al. [83] and Chen [84].

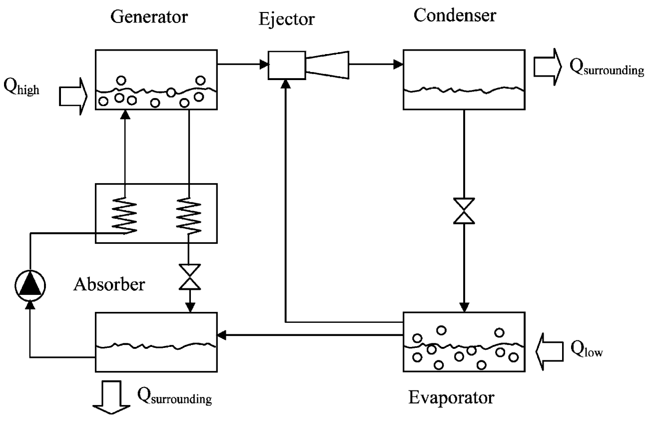


Fig. 15. Schematic diagram of the hybrid ejector-absorption cycle by Aphornratana and Eames [64].

Not only an increasing of absorber pressure, but an increasing of generator temperature can also improve the performance of single-effect absorption refrigeration machine [25]. One notable approach, proposed by Aphornratana and Eames [64], is shown in Fig. 15. An ejector is placed between a generator and a condenser. If a high-temperature heat source is available, the generator temperature may be raised and the solution concentration can be maintained constant. Therefore, the occurrence of crystallization in the generator is prevented. The ejector uses high pressure vapour refrigerant from the generator as the motive fluid. At the evaporator, the vapour refrigerant is not only absorbed into solution in the absorber entrained but also entrained by the ejector. Thus, the COP is higher than that for a conventional system. Experimental investigation showed that COP of 0.86–1.04 could be achieved. The major drawback of this system is, the generator must be operated at very high temperature (190–210 °C), and therefore, a corrosion of material may be problematic. The mathematical description of this system was given later by Sun et al. [30]. In the same manner as Sun et al., the simulation of Alexis and Rogdakis [10] was done on a similar system using CH_4O/H_2O as the refrigerant. In their study, solar energy was assumed to be the energy source of the refrigerator.

The approach proposed by the series of studies of Eames and Wu [47,65,86] is shown in Fig. 16. This cycle is a combined cycle between a steam jet heat pump and a single-effect absorption refrigerator. The steam jet heat pump is used to recover heat rejected during the condensation of the refrigerant. This recovered heat is supplied back to the generator of an absorption system. In this case, the maximum temperature of the solution is 80 °C. Therefore, the corrosion problem is eliminated. The experimental COP of 1.03 was claimed. Other studies related to hybrid ejector-absorption cycle were also presented in [8,9,87].

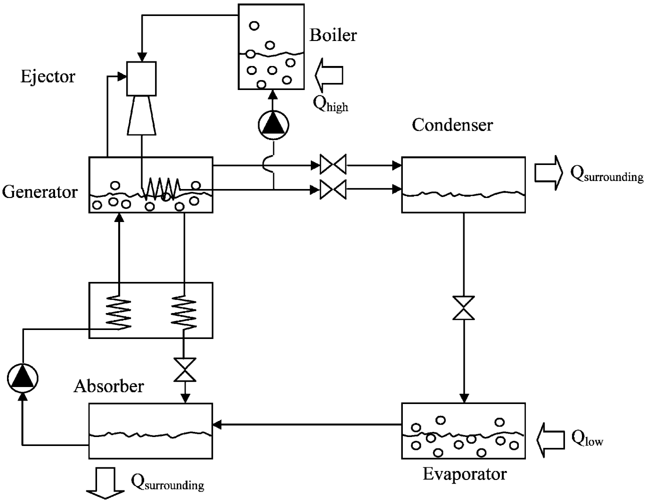


Fig. 16. Schematic diagram of the hybrid ejector-absorption cycle by Eames and Wu [47,65,86].

7. Conclusion

This paper describes a basic background and development of an ejector and its application in refrigeration purposes. At this moment, it can be said that the understanding in ejector theory has not been completely cleared. New assumptions on mixing and flowing characteristic were always established and applied on the mathematical model and computer simulation analysis. Even though these simulated results were claimed to become more accurate than others, very few of them were experimentally verified and approved.

An ejector is the critical component of a jet refrigeration system. Not only the system operating conditions, but two parameters, used to express the system performance (entrainment ratio and critical pressure), were also found to directly depend on ejector geometries and its working fluid. Many recent studies proposed and tested some new criterion in designing an ejector which has higher pressure lift performance. These ideas were to minimize any losses created by a mixing and shocking process. Halocarbon refrigerants seem to be a practical and appropriate working fluid for jet refrigeration system. Compared to the water system, the halocarbon refrigerator can provide higher performance and the required heat source temperature is lower. Therefore, the low grade energy, such as a solar energy, can be utilized and drive the system. Even though, the jet refrigerator has suffered from its very low COP, the improved COP from combining ejector to other types of refrigeration system (vapour compression and absorption system) was remarkable.

It is hoped that this contribution will stimulate wider interest in the technology of ejectors and their applications in refrigeration system. It should be useful for any newcomer in this field of technology.

Acknowledgements

The authors would like to thanks the Royal Golden Jubilee Program of Thailand Research Fund (TRF) for their financial support.

References

[1] Sun DW, Eames IW. Recent developments in the design theories and applications of ejectors—a review. J Inst Energy 1995;68(475):65–79.
[2] Zhang B, Shen S. Development of solar ejector refrigeration system. In: 1st International Conference on Sustainable Energy Technologies; 2002.
[3] Gosney WB. Principle of refrigeration. Cambridge: Cambridge University Press; 1982.
[4] Stoecker WF. Steam-jet refrigeration. Boston, MA: McGraw-Hill; 1958.
[5] Steam-jet refrigeration equipment, ch. 13. In: ASHRAE equipment handbook 1983: 13.1–13.6.
[6] Al-Khalidy N. Performance of solar refrigerant ejector refrigerating machine. ASHRAE Trans 1997;103(1):56–64.
[7] Rogdakis ED, Alexis GK. Design and parametric investigation of an ejector in an air-conditioning system. Appl Therm Engng 2000;20:213–26.
[8] Rogdakis ED, Alexis GK. Investigation of ejector design at optimum operating condition. Energy Conver Mngmnt 2000;41:1841–9.
[9] Alexis GK, Rogdakis ED. Performance characteristics of two combined ejector-absorption cycles. Appl Therm Engng 2002;22:97–106.
[10] Alexis GK, Rogdakis ED. Performance of solar driven methanol–water combined ejector-absorption cycle in the Athens area. Renew Energy 2002;25:249–66.
[11] Khattab NM, Barakat MH. Modeling the design and performance characteristics of solar steam-jet cooling for comfort air conditioning. Solar Energy 2002;73(4):257–67.
[12] Chen LT. Heat driven mobile refrigeration cycle analysis. Energy Conver 1978;18:25–9.
[13] Munday JT, Bagster DF. A new theory applied to steam jet refrigeration. Ind Eng Chem Process Des Dev 1977;16(4):442–9.
[14] Huang BJ, Jiang CB, Hu FL. Ejector performance characteristics and design analysis of jet refrigeration system. Trans ASME 1985;107:792–802.
[15] Huang BJ, Chang JM. Empirical correlation of ejector design. Int J Refrig 1999;22:379–88.
[16] Zeren F. Freon-12 vapor compression jet pump solar cooling system. PhD thesis, Texas A&M University; 1982.
[17] Eames IW, Wu S, Worall M, Aphornratana S. An experimental investigation of steam ejectors for application in jet-pump refrigerators powered by low-grade heat. Proc Instn Mech Eng A 1999;213:351–61.
[18] Chunnanond K, Aphornratana S. An experimental investigation of steam-ejector refrigerator, the analysis of pressure profile along ejector. In: The Asia-Pacific Conference on Sustainable Energy and Environment Technologies; 2003, pp.184–8.
[19] Chen YM, Sun CY. Experimental study of the performance characteristics of a steam-ejector refrigeration system. Exp Fluid Sci 1997;15:384–94.
[20] Keenan JH, Neumann EP, Lustwerk F. An investigation of ejector design by analysis and experiment. ASME J Appl MechTrans 1950;72:299–309.
[21] Sokolov M, Hershgal D. Enhanced ejector refrigeration cycles powered by low grade heat. Part 1. Systems characterization. Int J Refrig 1990;13:351–6.
[22] Sokolov M, Hershgal D. Enhanced ejector refrigeration cycles powered by low grade heat. Part 2. Design procedures. Int J Refrig 1990;13:357–63.
[23] Sokolov M, Hershgal D. Enhanced ejector refrigeration cycles powered by low grade heat. Part 3. Experimental results. Int J Refrig 1990;14:24–31.
[24] Abdel-Aal HK, Al-Zakri AS, El-Sarha ME, El-Swify ME, Assassa GM. Other options of mass and energy input for steam jet refrigeration systems. Chem Engng J 1990;45:99–110.

[25] Aphornratana S. Theoretical and experimental investigation of a combine ejector-absorption refrigerator. PhD thesis, University of Sheffield, UK; 1994.
[26] Sun DW. Variable geometry ejectors and their applications in ejector refrigeration systems. Energy 1996;21(10):919–29.
[27] Tomasak ML, Radermacher R. Analysis of a domestic refrigeration cycle with an ejector. ASHRAE Trans 101(45):1431–8.
[28] Eames IW, Aphornratana S, Haider H. A theoretical and experimental study of a small-scale steam jet refrigerator. Int J Refrig 1995;18(6):378–86.
[29] Sun DW. Solar powered combined ejector-vapour compression cycle for air conditioning and refrigeration. Energy Conver Mngmt 1997;38(5):479–791.
[30] Sun DW, Eames IW, Aphornratana S. Evaluation of a novel combined ejector-absorption refrigeration cycle—I: computer simulation. Int J Refrig 1996;19(3):172–80.
[31] Aly NH, Karameldin A, Shamloul MM. Modeling and simulation of steam jet ejectors. Desalination 1999;123:1–8.
[32] Riffat SB, Holt A. A novel heat pipe/ejector cooler. Appl Therm Engng 1998;18(3-4):93–101.
[33] Korres CJ, Papaioannou AT, Lygerou V, Koumoutsos NG. Solar cooling by thermal compression the dependence of the jet thermal compressor efficiency on the compression ratio. Energy 2002;27:795–805.
[34] Aphornratana S. Theoretical study of a steam-ejector refrigerator. RERIC Int Energy J 1996;18(1):61–74.
[35] Al-Khalidy N, Zayonia A. Design and experimental investigation of an ejector in an air-conditioning and refrigeration system. ASHRAE Trans 1995;101(2):383–91.
[36] Hoggarth ML. The design and performance of high-pressure injector as gas jet booster. Proc Instn Mech Eng 1970;185(56-71):755–66.
[37] Huang BJ, Chang JM, Wang CP, Patrenko VA. A 1-D analysis of ejector performance. Int J Refrig 1999;22:354–64.
[38] Cizungu K, Mani A, Groll M. Performance comparison of vapour jet refrigeration system with environment friendly working fluid. Appl Therm Engng 2001;21:585–98.
[39] Bevilacqua P. Mechanical refrigeration using and ejector-injector to transfer the working fluid from the evaporator to the condenser. Int J Refrig 1991;14:137–9.
[40] El-Dessouky H, Ettouney H, Alatiqi I, Al-Nuwaibit G. Evaluation of steam jet ejector. Chem Engng Process 2002;41:551–61.
[41] Zhang XJ, Wang RZ. A new combined adsorption-ejector refrigeration and heating hybrid system powered by solar energy. Appl Therm Engng 2002;22:1245–58.
[42] Rusly E, Aye Lu, Charters WWS, Ooi A, Pianthong K. Ejector CFD modeling with real gas model. In: Mechanical Engineering Network of Thailand the 16th Conference; 2002, pp. 150–5.
[43] Keenan JH, Neumann EP. A simple air ejector. ASME J Appl MechTrans 1942;64:75–81.
[44] Chou SK, Yang PR, Yap C. Maximum mass flow ratio due to secondary flow choking in an ejector refrigeration system. Int J Refrig 2001;24:486–99.
[45] Desevaux P. A method for visualizing the mixing zone between two co-axial flows in an ejector. Optics Lasers Engng 2001;35:317–23.
[46] Riffat SB, Gan G, Smith S. Computational fluid dynamics applied to ejector heat pumps. Appl Therm Engng 1996;16(4):291–7.
[47] Wu S, Eames IW. A novel absorption-recompression refrigeration cycle. Appl Therm Engng 1998;19:1149–57.
[48] Eames IW. A new prescription for the design of supersonic jet-pumps: the constant rate of momentum change method. Appl Therm Engng 2002;22:121–31.
[49] Chunnanond K, Aphornratana S. A study of steam ejector refrigeration cycle, effect of ejector geometries on ejector performance. In: The Second Regional Conference on Energy Technology towards a Clean Environment; 2000, pp. 390–7.
[50] Garris Jr CA. Pressure exchange ejector and refrigeration apparatus and method. United States Patent, 1997, patent number 5,647,221.

[51] Chang YJ, Chen YM. Enhancement of a steam-jet refrigerator using a novel application of the petal nozzle. Exp Therm Fluid Sci 2000;22:203–11.
[52] Sokolov MO. perational envelope and performance curves for a compression-enhanced ejector refrigeration system. ASHRAE Trans 1991;17(part 2):394–402.
[53] Sun DW, Eames IW. Performance characteristics of HCFC-123 ejector refrigeration cycles. Int J Energy Res 1996;20(10):871–85.
[54] Huang BJ, Chang JM, Petrenko VA, Zhuk KB. A solar ejector cooling system using refrigerant R141b. Solar Energy 1998;64(4-6):223–6.
[55] Huang BJ, Petrenko VA, Samofatov IY, Shchetinina NA. Collector selection for solar ejector cooling system. Solar Energy 2001;7(4):269–74.
[56] Sokolov M, Hearshgal D. Solar-powered compression enhanced ejector air conditioning. Solar Energy 1993;51(3):183–94.
[57] Wolpert JL, Riffat SB. 13 kW cooling hybrid solar/gas ejector air conditioning system. In: 1st International Conference on Sustainable Energy Technologies; 2002.
[58] Nguyen VM, Riffat SB, Doherty PS. Development of a solar-powered passive ejector cooling system. Appl Therm Engng 2001;21:157–68.
[59] Aphornratana S, Eames IW. A small capacity steam-ejector refrigerator: experimental investigation of a system using ejector with movable primary nozzle. Int J Refrig 1997;20(5):352–8.
[60] Wang J, Chen G, Jiang H. Study on a solar-driven ejection absorption refrigeration cycle. Int J Energy Res 1998;22:733–9.
[61] Chen SL, Yen JY, Huang MC. An experimental investigation of ejector performance based upon different refrigerants. ASHRAE Trans 1998;104(part 2):153–60.
[62] Eames IW, Aphornratana S, Sun DW. The jet-pump cycle-a low cost refrigerator option powered by waste heat. Heat Recovery Syst CHP 1995;15(8):711–21.
[63] Aphornratana S. Experimental investigation of a small capacity steam-ejector refrigerator. RERIC Int Energy J 1996;18(2):83–95.
[64] Aphornratana S, Eames IW. Experimental investigation of a combined ejector-absorption refrigerator. Int J Refrig 1998;22:195–207.
[65] Eames IW, Wu S. Experimental proof-of-concept testing of an innovative heat-powered vapour recompression–absorption refrigerator cycle. Appl Therm Engng 2000;20:721–36.
[66] Nahdi E, Champoussin JC, Hostache G, Cheron J. Optimal geometric parameters of a cooling ejector-compressor. Int J Refrig 1993;16(1):67–72.
[67] ESDU, Ejector and jet pump, data item 86030, London: ESDU International Ltd; 1985.
[68] Sokolov M, Hearshgal D. Optimal coupling and feasibility of a solar powered year-round ejector air conditioner. Solar Energy 1993;50(6):507–16.
[69] Grazzini G, Mariani A. A simple program to design a multi-stage jet pump for refrigeration cycles. Energy Conver Mngmnt 1998;39(16-18):1827–34.
[70] Al-Khalidy N. An experimental study of an ejector cycle refrigeration machine operating on R113. Int J Refrig 1998;21(8):617–25.
[71] Aphornratana S, Chungpaibulpatana S, Srikhirin P. Experimental investigation of an ejector refrigerator: effect of mixing chamber geometry on system performance. Int J Energy Res 2001;25:397–411.
[72] Sun DW. Comparative study of the performance of an ejector refrigeration cycle, operating with various refrigerants. Energy Conver Mngmnt 1999;40:873–84.
[73] Holton WC. Effect of molecular weight of entrained fluid on the performance of steam-jet ejector. ASME Trans 1951;October:905–10.
[74] Hamner RH. An alternate source of cooling: the ejector-compression heat pump. ASHRAE J 1980:62–6.
[75] Hamner RM. An investigation of an ejector-compression refrigeration cycle and its applications to heating, cooling and energy conservation. PhD thesis, The University of Alabama, Birmingham; 1978.
[76] Huang BJ, Petrenko VA, Chang JM, Lin CP, Hu SS. A combined-cycle refrigeration system using ejector cooling cycle as the bottom cycle. Int J Refrig 2001;24:391–9.

[77] Dorantes R, Estrada CA, Pilatowsky I. Mathematical simulation of a solar ejector-compression refrigeration system. Appl Therm Engng 1996;16(8-9):669–75.
[78] Al-Khalidy N. Experimental investigation of solar concentrators in a refrigerant ejector refrigeration machine. Int J Energy Res 1997;21:1123–31.
[79] Abdullah K, Syauful M. Development of solar powered jet cooling machine. Asian J Sci Technol 1997:153–160.
[80] Gokthun S. Optimazation of irreversible solar assisted ejector-vapor compression cascaded system. Energy Conver Mngmnt 2000;41:625–31.
[81] Srikhirin P, Aphornratana S, Chungpaibulpatana S. A review of absorption refrigeration technologies. Renew Sustain Energy Rev 2001;5:343–72.
[82] Kuhlenschmidt D. Absorption refrigeration system with multiple generator stages, US patent no. 3717007.
[83] Chung H, Huor MH, Prevost M, Bugarel R. Domestic heating application of anabsorption heat pump, directly fired heat pump. In: Proceedings International Conference University of Bristol, 19–20 September;1984.
[84] Chen LT. A new ejector-absorber cycle to improve the COP of an absorption system. Appl Energy 1998;30:37–41.
[85] Shi L, Yin J, Wang X, Zhu MS. Study on a new ejection-absorption heat transformer. Appl Energy 2001;68:161–71.
[86] Eames IW, Wu S. A theoretical study of an innovative ejector powered absorption-recompression cycle refrigerator. Int J Refrig 2000;23:475–84.
[87] Goktun S. Optical performance of a combined absorption and ejector refrigerator. Energy Conver Mngmnt 1999;40:51–8.